DUAL DRIVE ACTUATORS

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ABSTRACT

A new class of electromechanical actuators is described. These dual drive actuators were developed for the NASA-JPL Galileo Spacecraft. The dual drive actuators are fully redundant and therefore have high inherent reliability. They can be used for a variety of tasks, and they can be fabricated quickly and economically.

INTRODUCTION

A new class of electromechanical actuators has been developed for the NASA-JPL Galileo Spacecraft. These actuators perform such diverse functions as deployment of the 4.8-meter-high gain antenna, deployment and pointing control for the 1.0-meter probe relay antenna, and activation of a variable spring rate device in the spacecraft nutation damping system.

The actuators are called dual drives (Figures 1 and 2). They provide two independent electromechanical drive trains that combine at a common output shaft. Both trains are continuously engaged and independently operable without common failure modes.

The dual drive motor is a brushless configuration, containing internal electronics that perform all commutation functions, thereby providing a "two-wire" electrical interface and a motor package having speed/torque/current/weight characteristics equivalent to a standard size 11, permanent magnet brush motor. The dual drive configuration can also be adapted to use typical aerospace motor types, such as brush motors and stepper motors.

These are all desirable features. But, of equal importance, the dual drive can be produced quickly and economically due to its modular construction, manufacturing simplicity, and usage of commercial parts.

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OPERATING SEQUENCE

Figure 3(a) shows a mechanical schematic of the dual drive. The actual operating sequence is as follows:

System 1

Motor 1 drives spur gear 1 which rotates the input shaft of harmonic drive 1. The harmonic drive is a simple compact gear system for achieving large speed reductions and large output torques. The "pancake" harmonic drive, used here, includes four main elements as shown in Figure 3(b). The wave generator is the input member, and either circular spline can serve as the output member. Speed reduction is achieved by engagement of a differential number of teeth on the flex spline and stationary circular spline.

The input shaft to harmonic drive 1 passes through the center of a hollow outer shaft and continues through harmonic drive 2 without physical contact. The inner shaft is connected to the input of harmonic drive 1 through a cone clutch. The clutch is non-functional during normal operation, and its specific purpose will be discussed in a subsequent paragraph.

System 2

Motor 2 drives spur gear 2 which rotates the input shaft to harmonic drive 2. That input shaft is located concentrically around the system 1 drive shaft. The input drive elements of system 1 and system 2 are totally separate and non-contacting. However, operation of system 2 causes the entire system 1 harmonic drive to rotate as a single mass. The torque produced by system 2 is transmitted across harmonic drive 1 by the tooth mesh of flexspline 1.

SLIP CLUTCH FUNCTION

The system 1 harmonic drive is linked mechanically to ground through its wave generator, input gearing, and motor; therefore, relative motion must occur at one or more of these system 1 elements when system 2 is operated (system 2 must be able to rotate the entire system 1 harmonic drive as a single mass).

The required relative motion can occur by rotation of the system 1 wave generator bearing or by backdriving of the system 1 motor; however, when system 1 is not backdrivable, the only remaining point of relative motion will be the system 1 wave generator bearing. Therefore, failure of that bearing represents a potential functional single-point failure because both system 1 and system 2 would become non-operable.

A slip clutch, located within the system 1 input shaft, provides a redundant point of relative motion in the system 1 drive train. The clutch guarantees that regardless of input gear configuration, no functional single-point failures exist in the dual drive. (Functional single-point failures are discussed in additional detail in the paragraph dealing with dual drive reliability.)

SLIP CLUTCH TORQUE MARGIN

The slip clutch must be truly non-functional during normal operation in order to assure that it does not, by itself, become a single-point failure. This is accomplished by providing a very large torque margin between actual operating torques and the threshold torque for clutch slippage.

This arrangement guarantees that the clutch will be non-functional during system 1 operation and also guarantees that relative motion, during system 2 operation, will occur first at the system 1 wave generator bearing and/or motor. The magnitude of the slip clutch torque margin guarantees that the clutch will only operate if all other potential operating modes have failed.

Figure 4 shows the range of input shaft torques for the drive shown in Figure 1. The torque required to slip the clutch is supplied from system 2 output torque, but the clutch must only transmit input torque without slippage.

The ratio of input torque to output torque is, in fact, the harmonic drive numerical gear ratio times an efficiency factor. Thus, the torque required to slip the clutch only slightly reduces available output torque from system 2 while still assuring a significant input torque margin for system 1 operation.

BACKDRIVING

The term "backdriving" refers to the condition in which a torque, applied at the output shaft of a non-operating electromechanical drive, will cause rotation of the unit's input shaft and motor.

The dual drive model, shown in Figures 1 and 5, is a highly efficient low ratio configuration which can be backdriven. Each motor in the drive contains a specified magnetic detent (holding) torque. These detent torques, reflected through each gear system, provide a repeatable and predictable backdrive threshold of approximately 30 inch-pounds.

The operation of either dual drive system produces torque at the unit's output shaft. Also, torques may exist at the output shaft due to externally generated loads. The existance of output torque, regardless of the source, will produce a simultaneous torque reaction across <u>both</u> harmonic drives from the output shaft back to the stationary structure.

This output torque reaction is the source of backdriving torque. For the dual drive configuration under discussion, the torque reaction will cause a non-operating system to begin backdriving at an output torque level of approximately 30 inch-pounds. However, each active drive system, in this dual drive configuration, is capable of producing approximately 50 inch-pounds of output torque, and operation of either drive system while the companion system is non-operating will result in backdriving of that non-operating system.

When backdriving begins, the output speed of the dual drive will decrease to zero as motion is lost into the non-energized system, and the driving system will become torque limited at the backdriving threshold torque level. When both drive systems are operated simultaneously, backdriving cannot occur and the full output torque capability can be produced.

The backdriving action does not damage either system and, in applications where torque limiting or manual over-ride are required, a backdrivable dual drive configuration can satisfy those requirements.

If backdriving must not occur for specific applications, worm input gears should be selected.

DIFFERENTIAL OUTPUT SPEED

Combined operation of both dual drive systems may increase output torque by preventing backdriving, but the output torques of each dual drive system are not additive at the common output shaft. Dual drive configurations containing non-backdrivable systems will produce **similar** output torque levels for single or dual system operations; however, the output speeds of each dual drive system are additive at the common output shaft.

A slight speed differential will exist between system 1 and system 2 because both harmonic drive inputs rotate at the same speed (assuming identical motor speeds) while the body of harmonic drive 1 rotates at the output speed of harmonic drive 2. This speed differential amounts to: $N/N \pm 1$, where (N) equals the harmonic drive gear ratio. Thus, the speed difference amounts to 1% or less and can be ignored for most applications.

DUAL DRIVE RELIABILITY

Each dual drive system (drive train) contains a minimum number of functional elements (elements involving relative motion). This alone results in high inherent reliability for each drive train. But, when these truly parallel paths are combined into a single operating system containing no common failure points, the resulting reliability is unmatched by more conventional, partially redundant drives which have at least a few common functional elements.

Conventional reliability analysis of a system containing series elements will show that the overall reliability of that system will always be less than the reliability of the least reliable, series element in the system.

The same analysis will also show that the reliability of a system containing truly parallel elements will be orders of magnitude greater than the reliability of the least reliable element in that system.

The following equations relate load, life, and reliability for ball bearings which are the usual common elements in partially redundant systems.

Life/reliability is shown by

$$\frac{L}{L_o} = \left[\frac{\ln 1/R}{\ln 1/R_o}\right]^{1/1.125} \tag{1}$$

Reliability (R)	Available Life (L) Revolutions
0.9	1,000,000
0.99	124,000
0.999	16,000
0.9999	2,000
0.99999	265

Life/load is shown by

$$L = L_{10} \left(\frac{C}{P}\right)^3 \tag{2}$$

C = basic (B-10) load ratings
P = actual operating load
L = allowable life for 0.90 reliability
L_{10} = (1) million revolutions.

These equations indicate that a factor of eight decrease in required operating life or a factor of two decrease of operating loads will increase reliability by one additional "9." This means that the very great reliability inherent in the dual drive concept can be traded off against longer operating life and/or higher operating loads with a final calculated reliability equal to or better than the reliability of more common differential drive concepts.

DESIGN DETAILS

Figure 5 shows a cross-sectional view of one dual drive configuration. The input bearings (1 and 2) are "DF" mounted in order to simplify alignments. The input gears 3 and 4 are placed near the optimum locations to minimize input shaft/wave generator run-out effects on tooth mesh. These features eliminate the need for special "oldham" type couplings between the input gears and harmonic drives.

All ball bearing housings are fabricated from titanium alloy to allow wide temperature range operations. The output bearings (5 and 6) are "x-type" four-point contact ball bearings which can transmit axial, radial, and moment loads without the need for bearing pairs.

The motor register diameter 7 is slightly eccentric to the motor output shaft 8 . This allows adjustment of the "motor pinion-to-input gear" running clearance and also allows input gear ratio changes without the need for gear case modifications. (The ratio changes are accomplished by varying the diameters of driving and driven gears while maintaining an approximately constant gear center distance.)

The dual drive is lubricated with two types of grease. The alloy steel components of the gear systems are grease plated with Bray Oil Company Braycote 3L-38RP which provides lubrication and inhibits corrosion. The smaller

corrosion-resistant input ball bearings and gears are lubricated with Braycote 3L-38-1 which is chemically identical to the "RP" material except it contains no rust inhibitor. These materials provide excellent wide temperature range performance and very low vacuum outgassing, such that ambient pressure operations and vacuum operations provide indistinguishable results with regard to post-test lubricant condition.

The dual drive motor is shown in Figure 6. The motor has been specifically developed for spacecraft applications and has the following significant design features:

(1)	Wound Stator	Provides optimum heat transfer
(2)	Separate Magnetic Detent Assembly	Allows independent control of detent torque magnitude and number of detent poles.
(3)	Rare Earth Magnets	Minimize performance changes due to transient voltages, voltage reversals, or overspeeds due to external torques.
(4)	Steel Housing	Provides very low residual magnetic fields and wide temperature operating range.
(5)	Non-Contacting Rotor Position Sensor and Internal Drive Electronics	Provide brushless performance with simple "two-wire" electrical interface

DUAL DRIVE PERFORMANCE

Several dual drive configurations have been tested extensively to identify operating characteristics. The various test configurations are identified as follows:

(Model Number)	Motor Size		Input Gear Ratio & Type		Output Gear Type		Output Gear Ratio
	XX	-	XXX	-	XXX	/	XX

Example:

	11	-	10S	-	14 F	/	88
	Size 11 Motor		10 to 1 Spur Input Gears		Size 14 HDUF Harmonic Drive		88 TO 1 Harmonic Drive Ratio

(Overall Ratio = 10 x 88 = 880:1)

	11	-	30W	-	14 D	/	10,200
	Size 11 Motor		30 to Worm Input Gears		Size 14 HDD Harmonic Drive		10,200 to 1 Harmonic Drive Ratio

(Overall Ratio = 30 x 10,200 = 306,000:1)

Figure 7 shows general performance characteristics of three tested configurations. Figure 8 shows the speed/torque/current characteristics of these same configurations. Figure 9 shows the effects of temperature on speed and torque.

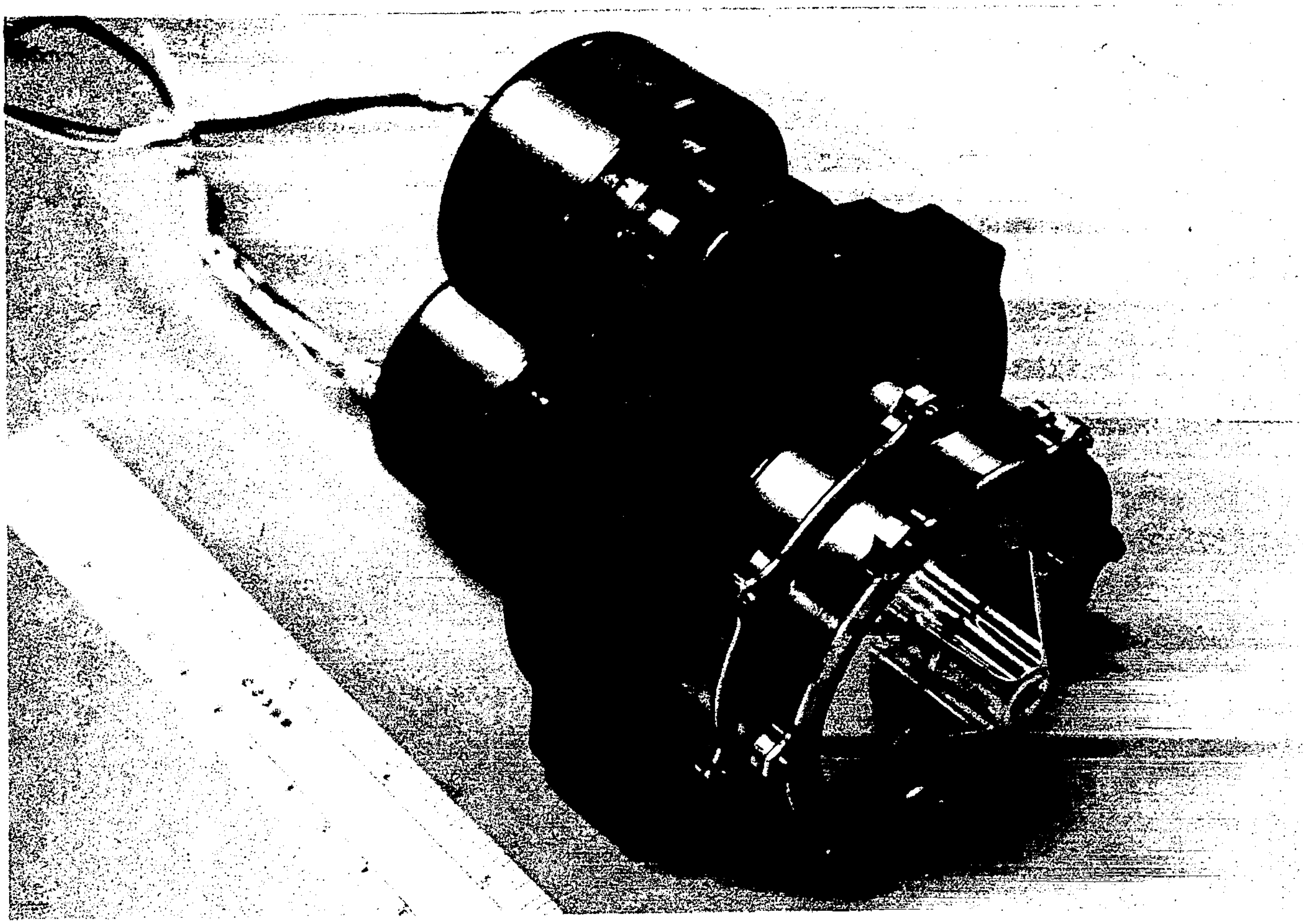
ALTERNATIVE DESIGN CONFIGURATIONS

The dual drive design provides for a modular construction in order to facilitate alternative design configurations. This concept is shown in Figure 10 along with estimated weights for several alternate design sizes. Figure 10 shows that approximately 40% of the component parts in the JPL dual drives are interchangeable between configurations.

Interchangeable spur gears or worm gears may be used at the input gear stages. These gears may then be coupled into any one of several available pancake harmonic drive configurations.

The standard pancake harmonic drives are produced in two configurations, "HDUF" (standard duty) and "HDD" (heavy duty). Approximately 127 different sizes and ratios are immediately available or can be readily produced.

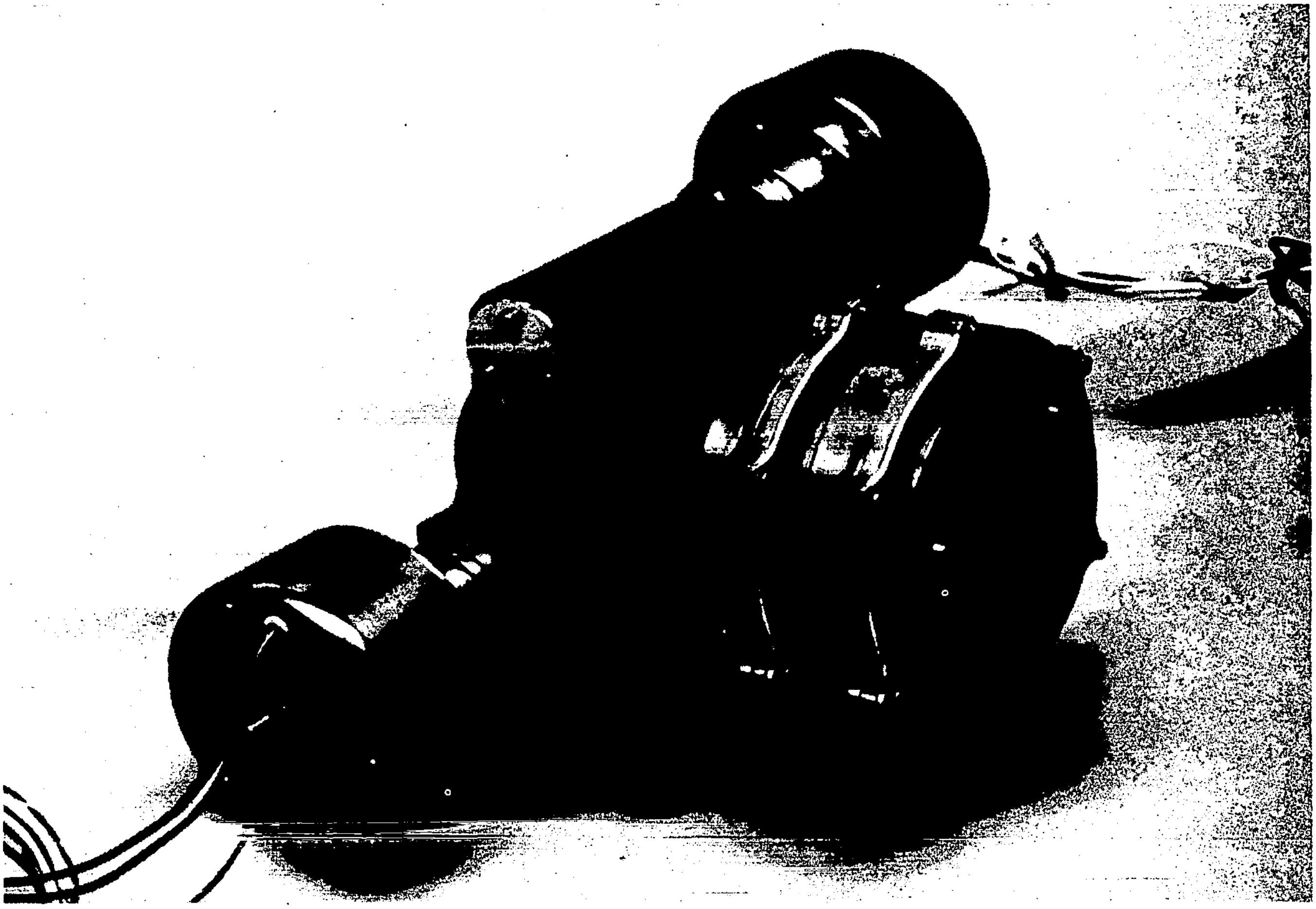
As shown in Figure 11, these gearing combinations make it possible to achieve overall gear ratios as low as 432:1 or as high as 612,000:1 within the same basic envelope. This allows great freedom in the selection of operating characteristics for specific user requirements, and significant component interchangeability is also maintained.



PERFORMANCE SPECIFICATION:

OUTPUT SPEED	0 - 13.4 RPM
OUTPUT TORQUE	0 - 50 in-lb
OVERALL GEAR RATIO	880:1
NORMAL OPERATING VOLTAGE	24 - 30 VDC
ALLOWABLE VOLTAGE	0 - 36 VDC
POWER (PER MOTOR) AT 30 VDC	3 - 11 W
MASS	2.00 lb

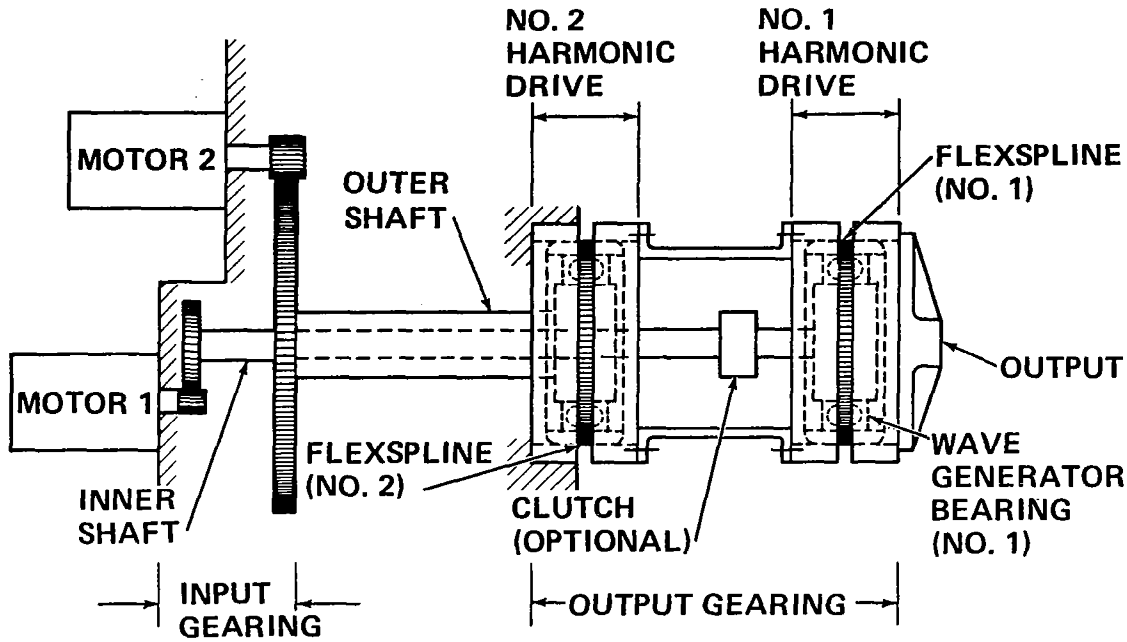
Figure 1. Dual Drive Assembly (JPL P/N 10095000)



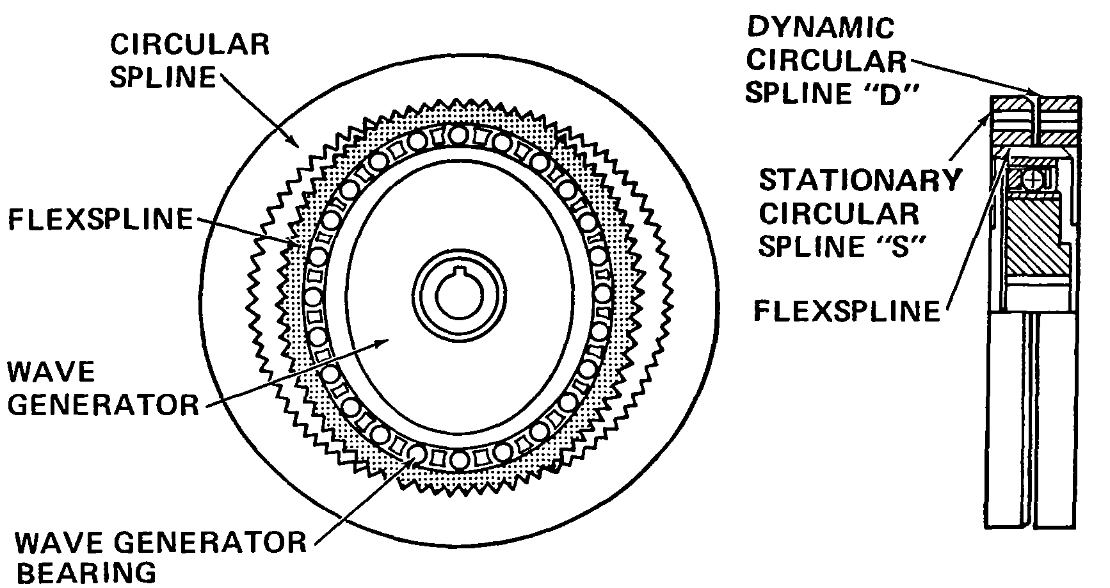
PERFORMANCE SPECIFICATION:

OUTPUT SPEED	0 - 0.018 RPM
OUTPUT TORQUE	0 - 180 in-lb
OVERALL GEAR RATIO	306,000:1
NORMAL OPERATING VOLTAGE	24 - 30 VDC
ALLOWABLE VOLTAGE	0 - 36 VDC
POWER (PER MOTOR) AT 30 VDC	4 - 11 W
MASS	2.50 lb

Figure 2. Dual Drive Assembly (JPL P/N 10100100)



(a) Overall



(b) Pancake Harmonic Drive
(Proprietary Product of the Harmonic Drive Div.,
USM Corp., Wakefield, MA)

Figure 3. Dual Drive Mechanical Schematic

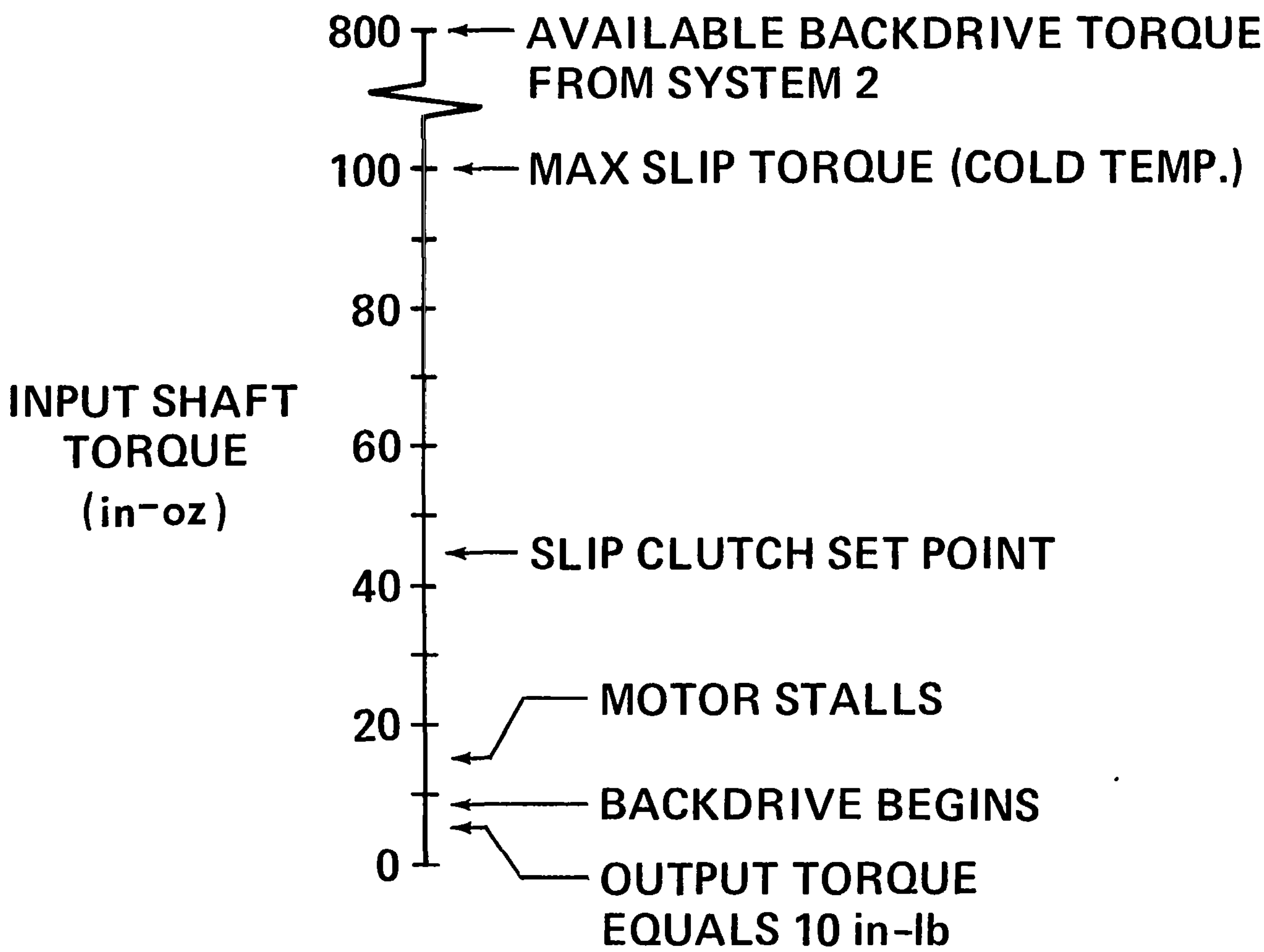


Figure 4. Input Shaft Torques (System 1)

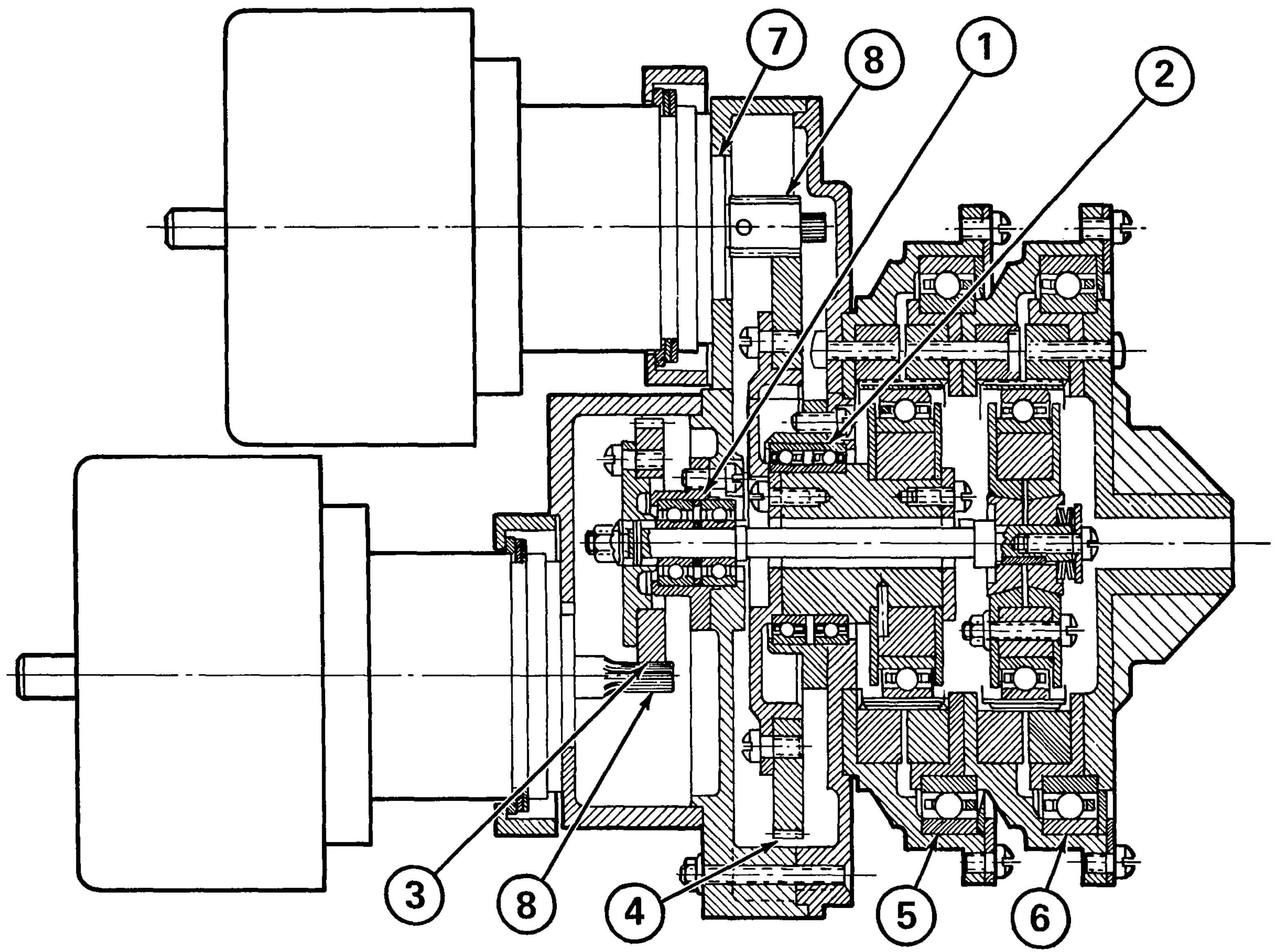
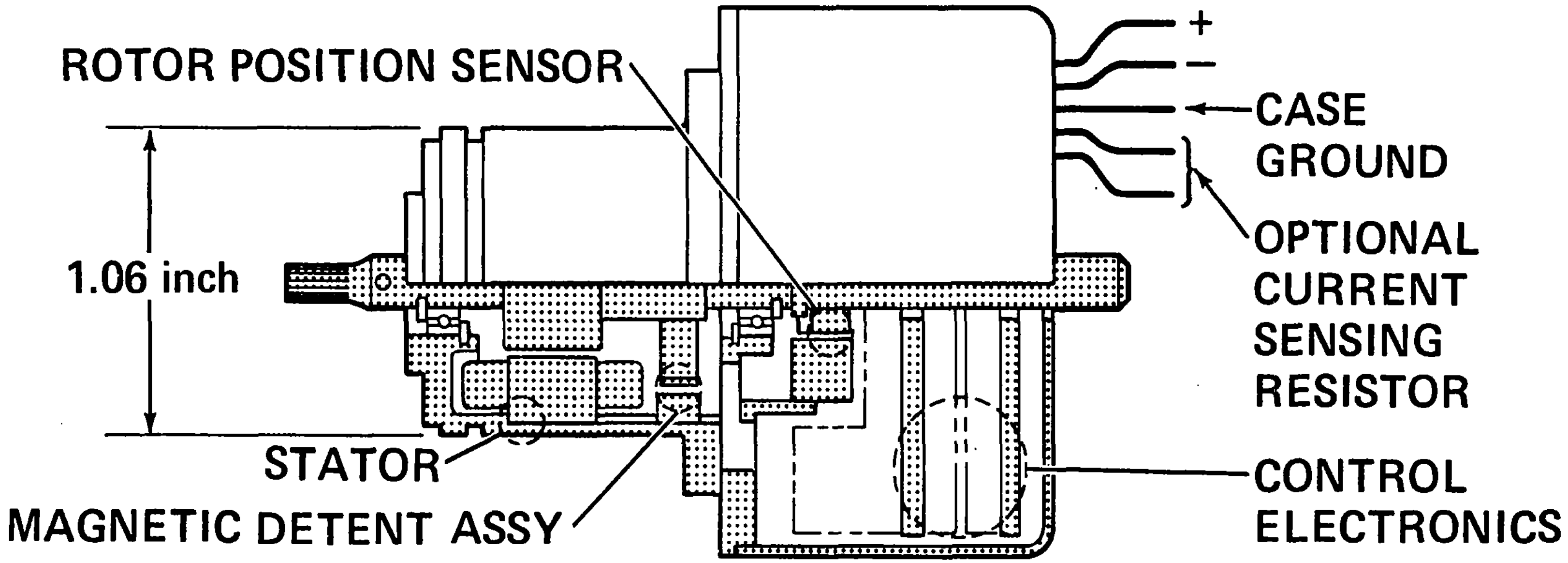


Figure 5. Dual Drive Assembly (Model 11-10S-14F/88)



MOTOR CHARACTERISTICS

NO LOAD SPEED	6800 RPM AT 30 V
STALL TORQUE	2.5 in-oz AT 30 V
NO LOAD CURRENT	0.05 A
DETENT TORQUE	0.5 in-oz
MASS	6.5 oz
RESIDUAL MAGNETIC FIELD	2 nanotesla AT 1.0 m
HOUSING	416 STAINLESS
MOTOR CONFIG	PM ROTOR/WOUND STATOR 3 ϕ, 4 POLE

Figure 6. Dual Drive Brushless D.C. Motor

PERFORMANCE CHARACTERISTIC		DUAL DRIVE MODEL NO. 11-10S-14F/88	DUAL DRIVE MODEL NO. 11-30W-14F/88	DUAL DRIVE MODEL NO. 11-30W-14D/10200
BACK LASH		2° OR LESS (STANDARD), 1° OR LESS (SPECIAL)		
OUTPUT SHAFT TORSIONAL WIND-UP		APPROXIMATELY 2000 in-lb/RADIAN		
OUTPUT MOTION LINEARITY		± 2 MIN (ONE SYSTEM OPERATING)		± 4 MIN (TWO SYSTEMS OPERATING)
OPERATING TEMPERATURE		QUALIFIED FOR -60°F TO $+160^{\circ}$F		
NON-OPERATING TEMPERATURE		QUALIFIED FOR -100°F TO $+160^{\circ}$F		
DEMONSTRATED LIFE		70,000 OUTPUT REVS (NO FAILURES)	IN TEST	IN TEST
OUTPUT BEARING ALLOWABLE LOADING	OPERATING	149 lb (RADIAL) 375 lb (THRUST) 168 in-lb (BENDING)	SAME	SAME
	NON-OPERATING	763 lb (RADIAL) 763 lb (THRUST) 231 in-lb (BENDING)	SAME	SAME
STATIC HOLDING TORQUE		30 in-lb (HOLDING) 20 in-lb (STOP AND HOLD)	GREATER THAN 50 in-lb	NON-BACKDRIVABLE
STALL IN A VACUUM		REQUIRED: 2 MINUTES AT 30 VDC AND AMB TEMP ENV DEMONSTRATED: 11 MINUTES AT 30 VDC AND AMB TEMP ENV		

Figure 7. Dual Drive General Performance Characteristics

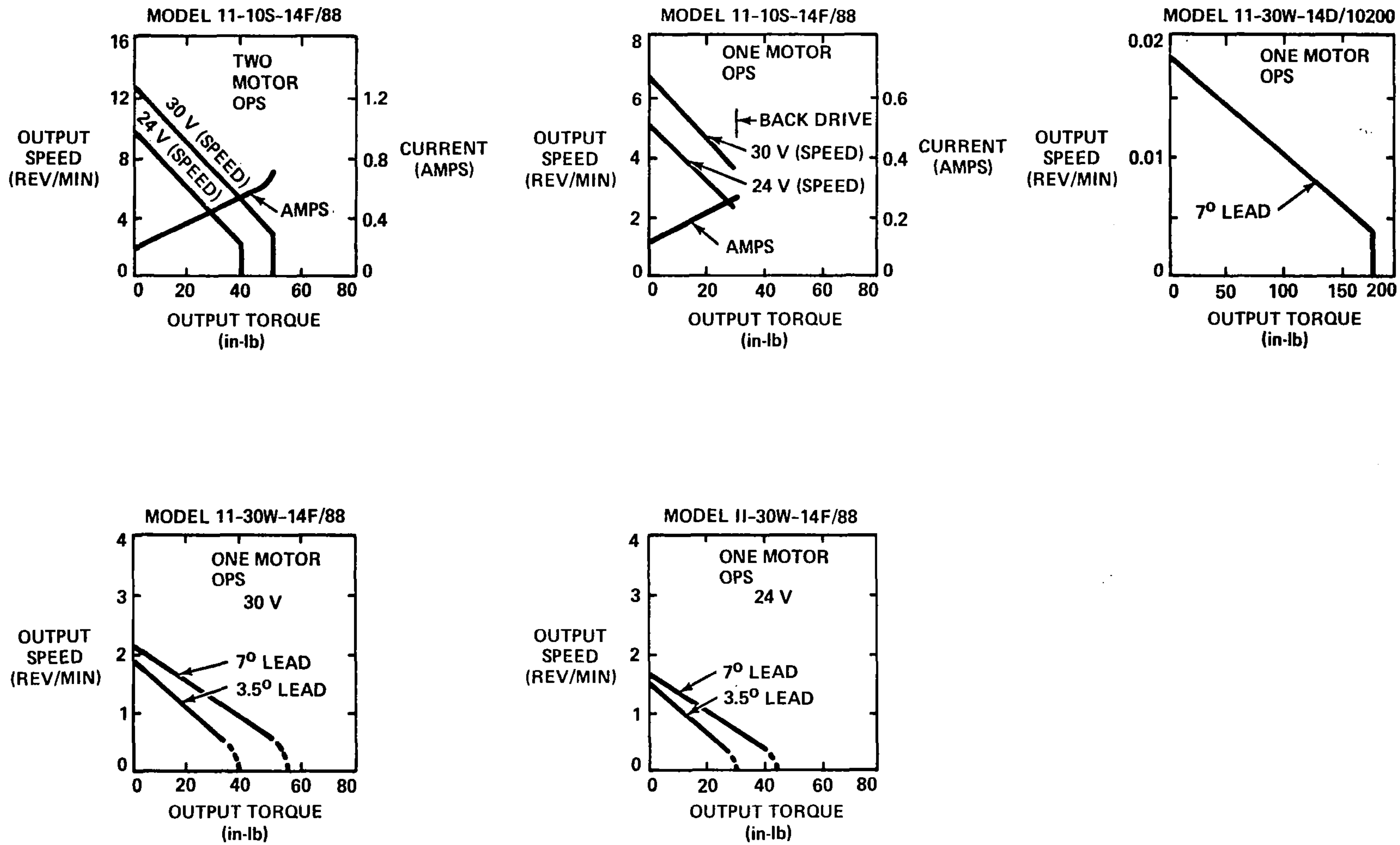


Figure 8. Speed/Torque/Current for Various Dual Drives

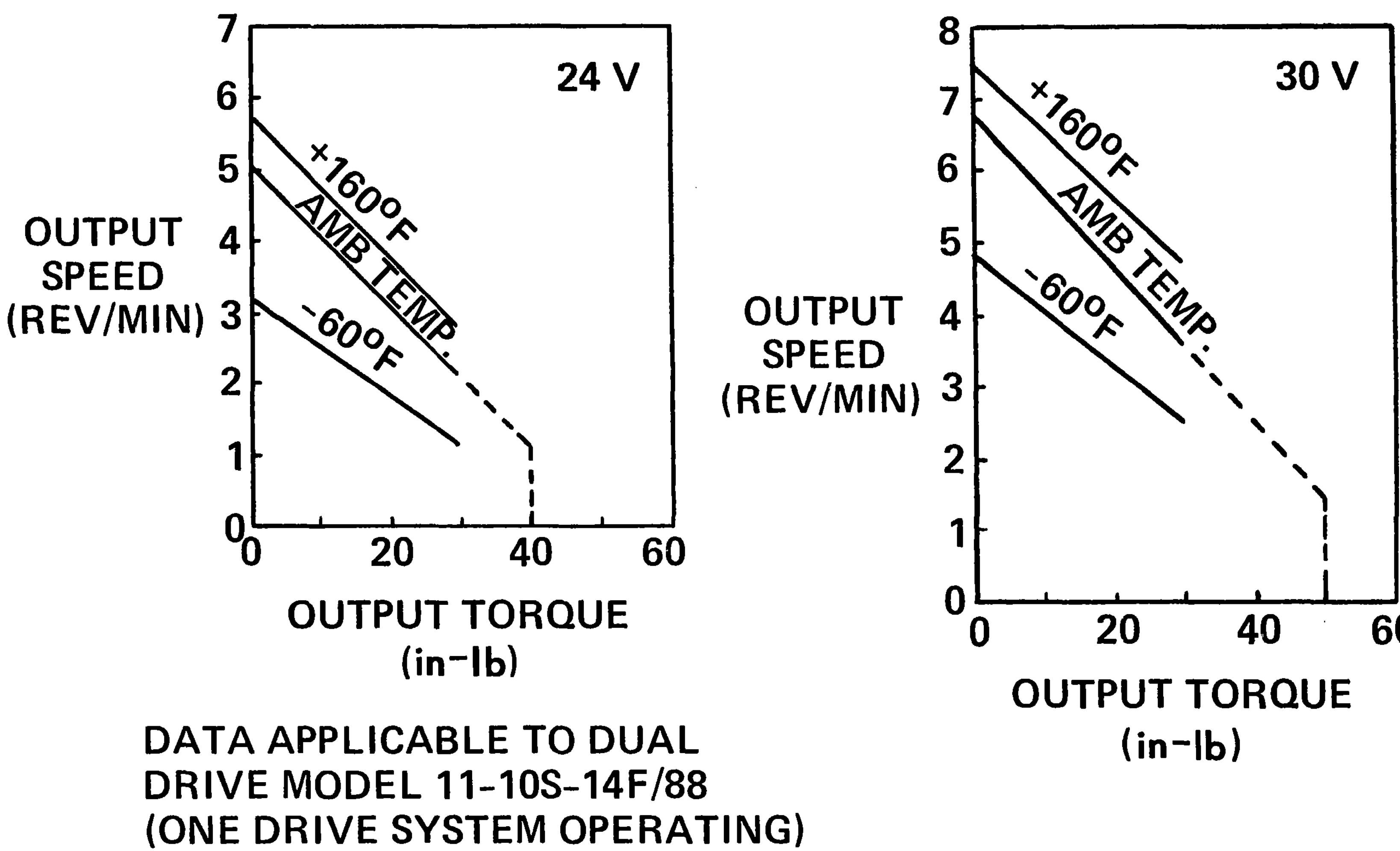
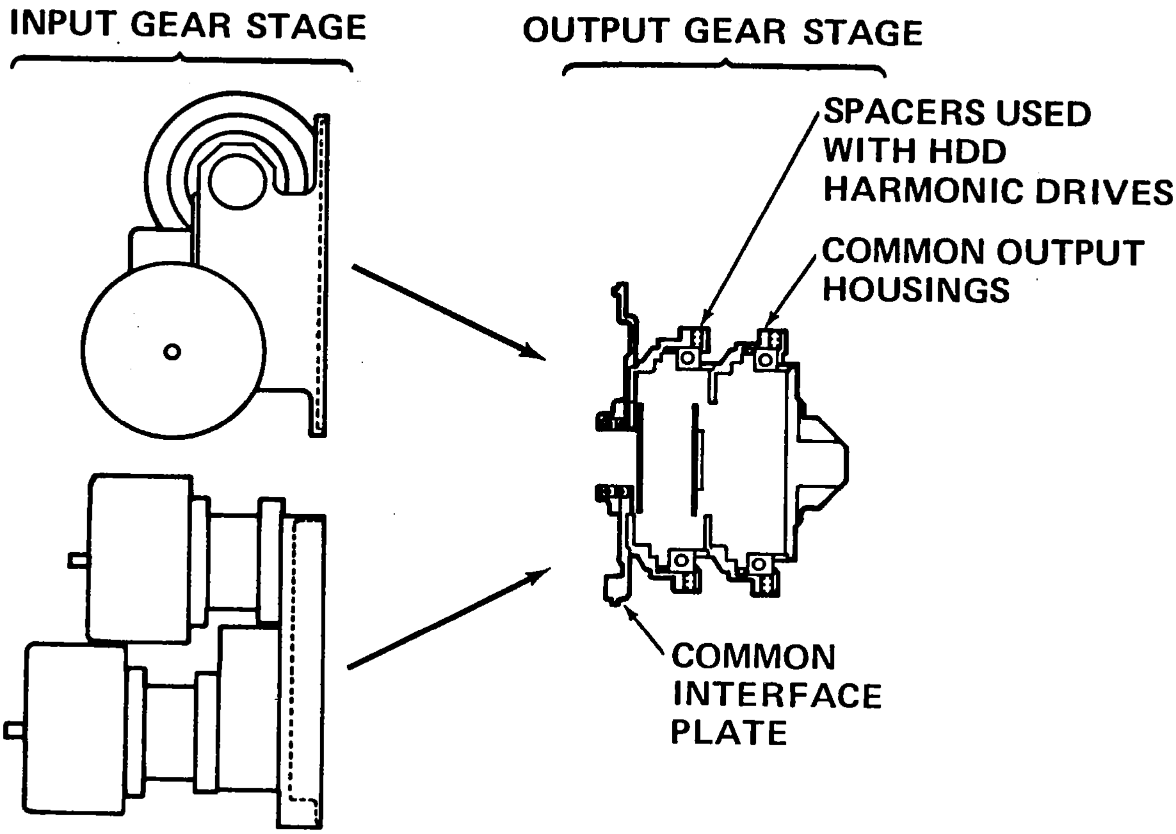


Figure 9. Temperature Effects on Dual Drive Performance



APPROXIMATE WEIGHT, lb

	OUTPUT GEARING						
INPUT CONFIG	14F	14D	20F	20D	25F	25D	32D
11-30 11-60	2.15	2.65	3.6	4.5	4.95	6.6	9.3
11-10	2.0	2.5	3.4	4.3	4.8	6.4	9.1
15-30 15-60		3.3	4.2	5.1	5.6	7.2	9.9
15-10		3.1	4.0	5.0	5.4	7.0	9.7

Figure 10. Dual Drive Modular Construction and Weight Estimate for Alternative Configurations

ALTERNATIVE DUAL DRIVE GEAR RATIOS (SIZE 14)*											
INPUT GEAR RATIOS	OUTPUT GEAR RATIO										
	SIZE 14F		SIZE 14D								
	88	110	72	80	100	110	296	489	1110	2331	10,200
6:1 SPUR	528	660	432	480	600	660	1776	2994	6660	13,986	61,200
10:1 SPUR	880	1100	720	800	1000	1100	2960	4990	11,100	23,310	102,000
30:1 SPUR	2640	3300	2160	2400	3000	3300	8800	14,970	33,300	69,930	306,000
30:1 WORM	2640	3300	2160	2400	3000	3300	8880	14,970	33,300	69,930	306,000
60:1 WORM	5280	6600	4320	4800	6000	6600	17,760	29,940	66,000	139,860	612,000

*SIMILAR RATIO COMBINATIONS APPLY TO OTHER HARMONIC DRIVE SIZES (20, 25, 32, 40, 50, 65, 80, 100)

Figure 11. Alternative Dual Drives

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Mr. Packard has been a member of the Technical Staff at Jet Propulsion Laboratory since 1978. Prior to that time, he was with Lockheed Missiles and Space Company in Sunnyvale. His experience has encompassed design and development involving over 300 spacecraft. Particular areas of knowledge include: deployable structures, pointing control systems, deployment devices, general-purpose space actuators, and pyrotechnic separation systems. Mr. Packard received his B.S. degree in Aeronautical Engineering from California State Polytechnic College in 1960.